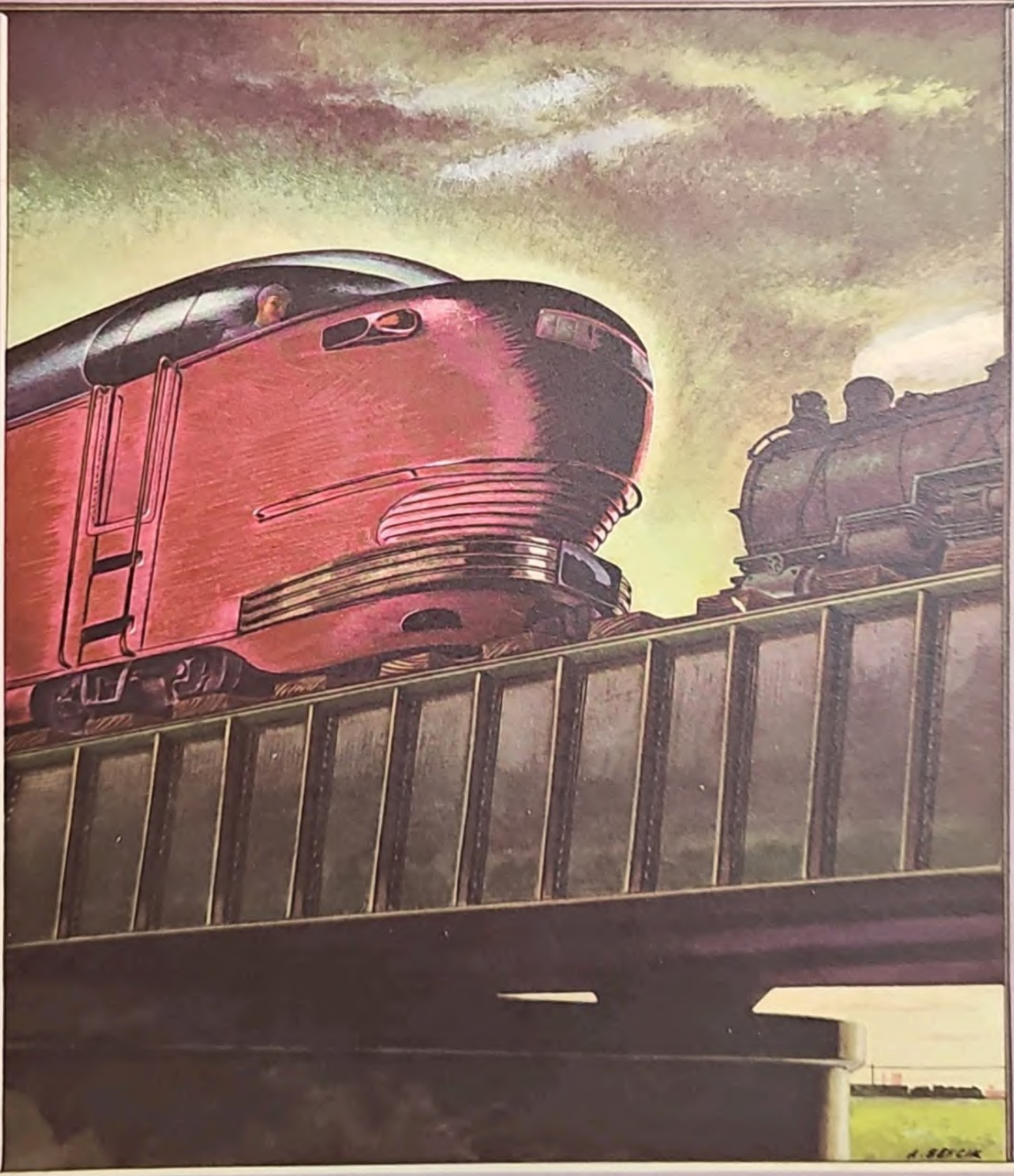
FASTENERS

VITAL IN EVERY INDUSTRY

PUBLISHED BY AMERICAN INSTITUTE OF BOLT, NUT, AND RIVET MANUFACTURERS



VOLUME ONE • NUMBER FOUR

AMERICAN INSTITUTE OF BOLT, NUT AND RIVET MANUFACTURERS

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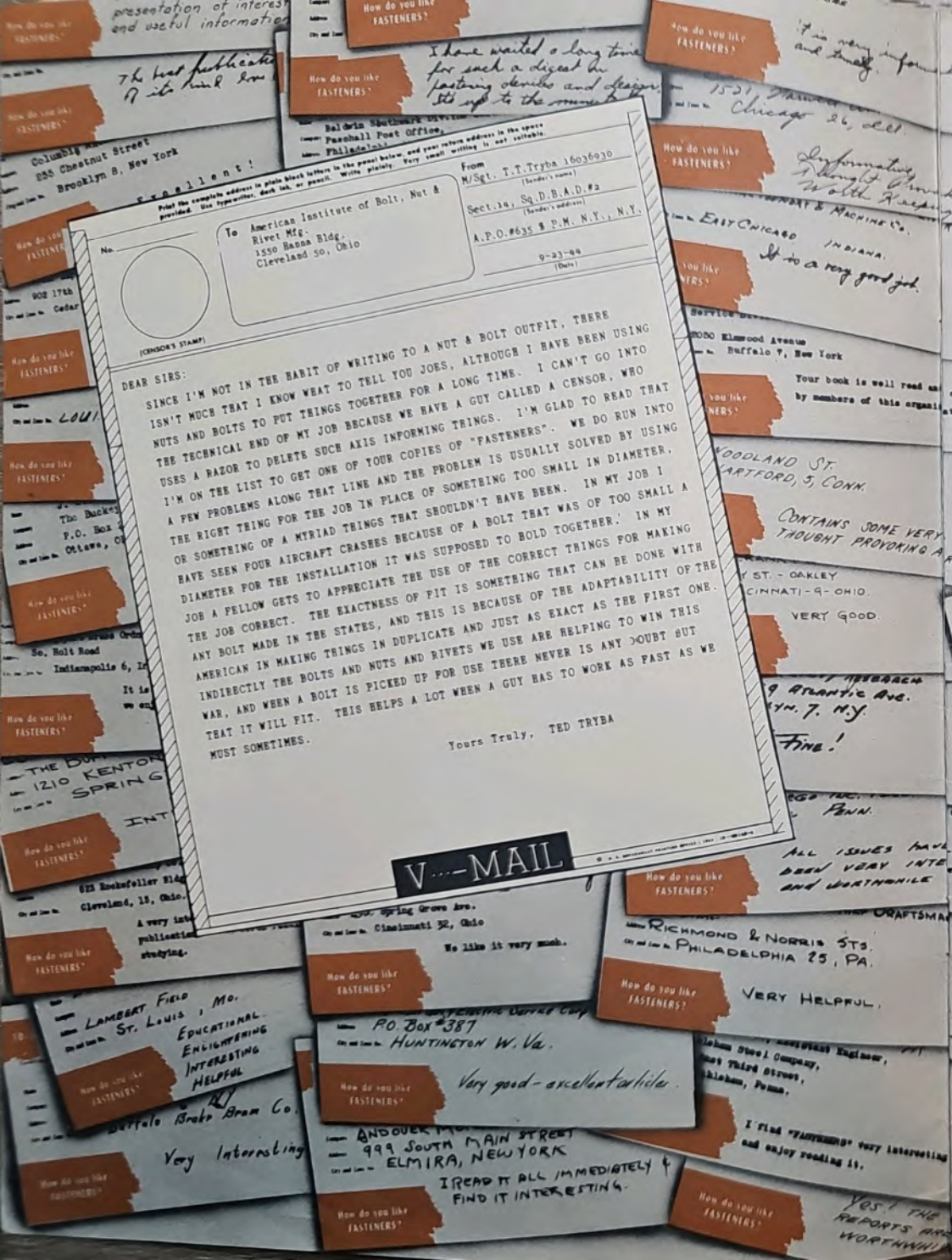
We are deeply appreciative for the fine response to our requests in the last issue of "Fasteners" asking for suggestions as to how the usefulness of this publication might be increased.

The replies have run into the thousands, and an analysis of them indicates that the suggestions will prove to be of great value in planning an editorial program of top interest to our readers. Future issues of "Fasteners" will reflect these suggestions.

One of the most interesting responses received was a V-mail letter (see opposite page) from a service man on airplane maintenance in a combat area.

Herman N. Lind
President

Examination of a Jap aircraft bolt from a Zero shows that the cut thread has a close pitch tolerance, and the nut fits well. In general, however, its quality is poor compared to American standards. The Jap bolt is not pointed and lacks a washer face under the head. Checking on the comparator shows that the hex is out of line. The cotter hole is drilled at a point which is inaccessible when the nut is in a normal position. The nut is crudely made—simply a hex slug drilled and tapped, with no chamfer or washer face.


How do you like FASTENERS?
Print the complete address in plain block letters in the panel below, and your return address in the space provided. Use typewriter, dark ink, or pencil. Write plainly. Very small writing is not suitable.
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(CENSOR'S STAMP)
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From M/Sgt. T.T.Tryba 16036030
(Sender's name)
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(Sender's address)
A.P.O.#635 % P.M. N.Y., N.Y.
9-23-44
(Date)
DEAR SIRS:
SINCE I'M NOT IN THE HABIT OF WRITING TO A NUT & BOLT OUTFIT, THERE ISN'T MUCH THAT I KNOW WHAT TO TELL YOU JOES, ALTHOUGH I HAVE BEEN USING NUTS AND BOLTS TO PUT THINGS TOGETHER FOR A LONG TIME. I CAN'T GO INTO THE TECHNICAL END OF MY JOB BECAUSE WE HAVE A GUY CALLED A CENSOR, WHO USES A RAZOR TO DELETE SUCH AXIS INFORMING THINGS. I'M GLAD TO READ THAT I'M ON THE LIST TO GET ONE OF YOUR COPIES OF "FASTENERS". WE DO RUN INTO A FEW PROBLEMS ALONG THAT LINE AND THE PROBLEM IS USUALLY SOLVED BY USING THE RIGHT THING FOR THE JOB IN PLACE OF SOMETHING TOO SMALL IN DIAMETER, OR SOMETHING OF A MYRIAD THINGS THAT SHOULDN'T HAVE BEEN. IN MY JOB I HAVE SEEN FOUR AIRCRAFT CRASHES BECAUSE OF A BOLT THAT WAS OF TOO SMALL A DIAMETER FOR THE INSTALLATION IT WAS SUPPOSED TO BOLD TOGETHER. IN MY JOB A FELLOW GETS TO APPRECIATE THE USE OF THE CORRECT THINGS FOR MAKING THE JOB CORRECT. THE EXACTNESS OF FIT IS SOMETHING THAT CAN BE DONE WITH ANY BOLT MADE IN THE STATES, AND THIS IS BECAUSE OF THE ADAPTABILITY OF THE AMERICAN IN MAKING THINGS IN DUPLICATE AND JUST AS EXACT AS THE FIRST ONE. INDIRECTLY THE BOLTS AND NUTS AND RIVETS WE USE ARE HELPING TO WIN THIS WAR, AND WHEN A BOLT IS PICKED UP FOR USE THERE NEVER IS ANY DOUBT BUT THAT IT WILL FIT. THIS HELPS A LOT WHEN A GUY HAS TO WORK AS FAST AS WE MUST SOMETIMES.
Yours Truly, TED TRYBA
V---MAIL
Excellent!
I have waited a long time for such a digest on fastening devices and design. It's right to the minute!
Columbia Nut & Bolt Co.
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Baldwin Locomotive Works
Philadelphia
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Postmaster, Philadelphia Post Office
Chicago 26, Ill.
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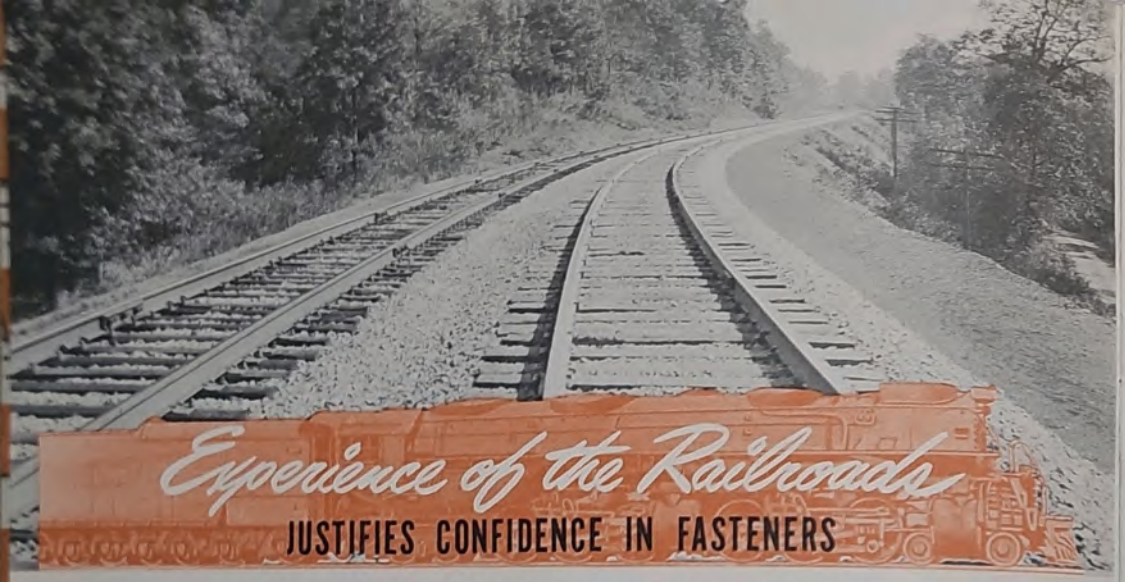
I READ IT ALL IMMEDIATELY & FIND IT INTERESTING.
I find "FASTENERS" very interesting and enjoy reading it.
YES! THE REPORTS ARE WORTHWHI



Experience of the Railroads JUSTIFIES CONFIDENCE IN FASTENERS

by WALTER S. LACHER, Secretary Engineering Division
Association of American Railroads

. . . two special applications of fasteners — the rail joint and the steel bridge — have been the subject of much engineering consideration . . . each of these applications is unique, with its own peculiar problems . . .

Committed from beginning to the extensive use of iron, the railroads still rank among the leading users of steel. It goes almost without saying, therefore, that they have always had a vital interest in metal fasteners. Omitting consideration of cars and locomotives, which occupy an enormous field all to themselves, there are two special applications of fasteners in which the railway industry maintains a distinctive position. One of these is in the railroad rail, a highly specialized structural element which imposes its own peculiar problems, and the other is the steel railroad bridge, which represents the evolution of a high order of engineering technique that is now applied as well to the steel frame building and highway bridges. Except for the fact that they comprise essential elements of the properties of the same industry, the rail and the bridge have nothing in common, and least of all in any problems presented by the fastenings provided for them. For this reason fastenings for rails and for bridges must be considered separately.

RIVETED STEEL BRIDGES

The history of the metal bridge, especially the steel bridge, is really the story of railway bridge engineering. Until the advent of modern highway construction for heavy bus and truck loads, the requirements for bridges for railways were so much more severe than those for roads and streets, that railway bridges monopolized the attention and stimulated the talents of the leading bridge engineers. It followed as a matter of course, therefore, that advances in the art of bridge building usually found their first expression in some railroad structure. Likewise, when the American Railway Engineering Association was organized in 1899, it was natural that its Committee on Iron and Steel Structures would assume leadership in developing and unifying practice in steel bridge design, a leadership that became so well established as to be recognized the world over and to serve as the starting point from which highway bridge engineers launched their own branch of the profession about a generation ago.

The steel frame structure is an assembly job. Because of limitations imposed by manufacture, transportation and hoisting equipment, it represents the result of the skillful shaping, fitting, assembling and connecting of a great many separate pieces of metal. Consequently, a large portion of the time of the bridge engineers has been devoted to problems bearing on the making and improving of connections involving the use of rivets, bolts and the pins of pin-connected structures. No better evidence of this is needed than the space devoted to fastenings in the AREA Specification for Steel Rail-

Walter S. Lacher

way Bridges. In the section on design, fully 22 per cent, and in the section on workmanship almost 33 per cent of the text matter is given over to stipulations relating to rivets, bolts and pins, the holes in which they are placed, their spacing and distribution, and other problems bearing on the general subject of connections.

So far as the materials are concerned a high degree of uniformity in practice has been realized; the AREA has adopted the American Standard for large rivets and its specifications for rivet steel are substantially the same as those of the American Society for Testing Materials. The requirements for workmanship embodied in the AREA specifications represent the result of a process of evolution in which the experiences of the railways and the fabricators have been pooled. Practice has become stabilized and no marked innovations are expected. There are still minor differences of opinion with respect to the need for reaming rivet holes, but the Association's stipulations on this point have not been subject to any appreciable modification for more than a decade.

The section of the specifications devoted to design is also the result of years of observation and experience. General principles and basic values have long been established and have stood the test of time. Given a particular set of conditions, established practice, guided by the specifications, points the way to the particular construction that will best meet the needs of the situation and will inspire confidence in the effectiveness and permanence of the resultant design.

But this does not mean that the drafters of the specifications are not looking for opportunities for improvement. As a matter of fact they are perfectionists almost to a fault. Experience is always pointing the way to minor modifications; currently presented results of experimentation throw new light on old problems, and modern testing equipment gives an insight into phenomena heretofore undisclosed. However, the actual changes resulting from such developments are usually of a minor nature. No fundamental changes have taken place in years, but the structural engineer is still on the lookout for improvements.

Bridges represent the result of the skillful connection of a great many separate pieces and bridge engineers have devoted much time to the proper application of fasteners in these structures.

In conclusion, therefore, the attitude of the railway bridge engineer toward the riveted structure is one of confidence in its behavior under all circumstances, confidence in the specifications for design, workmanship and materials, and confidence in the criteria applied in rating old and overloaded bridges. The behavior of riveted joint when subjected to excessive stresses is no minor factor in the creation of this confidence, for the fact that distress in riveted joints is manifested by the appearance of loose rivets has long been recognized as a valuable index in the administration of bridge maintenance.

THE RAIL JOINT

Few elements of the various component parts of the railway plant have been subject to more extensive experimentation or the source of more patents than the rail joint. Yet in spite of this, the rail joint of today is founded on the same basic principles and consists of the same basic elements that were present in the joint first applied to the T-rail, namely two bars that engage fishing surfaces on the top of the base and the underside of the head, these two bars being held in close contact with the rail by means of four or more threaded bolts.

Notwithstanding the false hopes aroused by hundreds of designs for which fantastic advantages were claimed, the search for improvements continues, but rather than efforts to evolve radical departures from the conventional design, the quest for something better is now directed toward a study of each of the elements that make up the complete joint with a view to determining the faults of each and how these may be overcome. This attitude is based on the realization that the present construction has one basic virtue—it works; it is admittedly far from perfect, but it represents the survival of the fittest in a service test that has embraced almost every conceivable idea of what a joint might be. With this point of view on the part of railway maintenance officers, it is not surprising that various studies of the rail joint are now in progress, embracing inquiries into such problems as the relative merits of a variety of designs and shapes of joint bars, the cause of fatigue failures in the bars, and the behavior of the bolts that hold the bars in place, that is to say—the track bolts.

Track bolts perform a rather complex set of functions. They must apply adequate pressure against the joint bars and still permit sufficient play to allow the rails to change in length with changes in temperature. But further than this, the bolts must continue to apply pressure to the joint bars in spite of the progressive tendency of the bars to move closer together as wear takes place in the contact surfaces.

Bolt tension and means of maintaining it thus becomes one of the major problems in the study of the rail joint. For a long time primary effort was directed toward the devising of locking mechanisms to keep the nuts from backing off the bolts, but with the realization that the real problem lay in the means for taking up the effect of wear on the contact surfaces, as explained above, attention was focused on means of maintaining tension in the bolts, for example, helical washers and other forms of springs.

Still another approach has been found in programs for periodic tightening of bolts, aided and made easier by the development of power bolters or bolt tighteners, equipped in more recent models with "kick-offs" or clutch releases intended to insure the application of a desired torque as a means of insuring uniform tension. But these various developments, while leading to a definite improvement in performance, served actually to complicate the problem, because they introduced new questions which remained unanswered. Furthermore, there were too many differences of opinion as to the desired result. What, for example, is the optimum bolt tension? How is it possible to find out if any

The rail joint, apparently a simple connection, has been the subject of extensive experimentation and the source of many patents.

specified tension is being obtained? At what rate will the bolt tension be lost during the interval between tightening?

RESEARCH INVESTIGATIONS

It was in an effort to learn the answers to these and other questions that the research staff of the Association of American Railroads, at the instance of the Committee on Track, instituted a series of practical tests with the cooperation of five different railways on as many stretches of track. By providing conical seats in two ends of the bolts of the selected joints it was possible to determine the tension in the bolts at any time with a yoke type extensometer for such purposes as the following:

To determine the amount of tension placed in the bolts with power and hand wrenches, as a check on practice and the accuracy of power wrench "kick-offs".

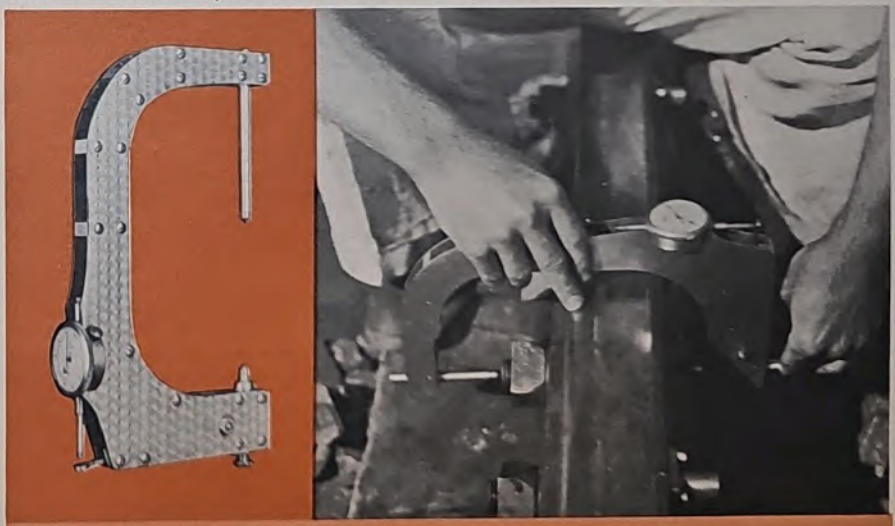
To determine loss of tension in terms of time or volume of traffic.

To determine the influence on bolt behavior of such variables as initial bolt tension, character of joint bars, kind of thread on the bolts, effect of the use and non-use of spring washers, etc.

Early in the tests it was discovered that there was much to be desired in the accuracy of the bolt tension values obtained by the kick-off mechanism of power operated wrenches, and further study disclosed the fact that this lack of accuracy was due only in part to limitations of the machines. Much of it was found to be the result of marked variations in the amount of torque required to turn the nuts on the bolts — in other words, differences in thread friction.

The latest report on these bolt tension measurements expresses a preference for bolts with a free thread fit, on the ground that they offer far greater uniformity of thread friction which, in turn, insures the attainment of greater uniformity in bolt tension. Whether this conclusion will result in any change in specifications cannot be determined at this time.

Other results of this investigation relate to practice in the maintenance of bolt tension, the initial tension to be applied, the amount to be applied in subsequent tightenings and the interval between tightenings. Reference is made also to the value of corrosion-resisting lubricants. Perhaps more important than any of these individual findings is the effect of a better understanding of the behavior of the bolt in performing its several functions as an essential element of the track joint, because it will bring about a better understanding between the producer and the user in any consideration of possible revisions of specifications.



Tension in track bolts may be measured by a yoke gauge, shown at left. The gauge is applied to the bolt in the rail joint as illustrated at right.

What Torque?

by W. C. STEWART

Technical Adviser, American Institute of Bolt, Nut and Rivet Manufacturers

. . . torque-tension ratio is an important factor in many applications of fasteners . . . friction gremlins, opposing efforts to produce tension, make this ratio an uncertain quantity . . .

RECENTLY a great deal of attention has been paid to the proper torquing of bolts, nuts, and screws. The American Institute of Bolt, Nut and Rivet Manufacturers has received so many requests for information as to the torque required to produce a desired initial tension in bolt or screw shanks that it is thought some general discussion would be useful even though no universal answer is as yet available.

The necessity of producing tension in bolts has long been known in industries such as the power piping industry, where sufficient tension must be set up in flange bolts, for example, to exceed the pressure in the piping or the flange will leak. More recently, J. O. Almen, of the Research Laboratories Division, General Motors Corporation, has pointed out the great advantage of initial bolt tension in improving the endurance life of dynamically loaded joints (See Fasteners Vol. 1, Nos. 1 and 2). This clamping tension is also of advantage in relatively statically loaded structures where rigidity of joints is required, since this clamping force, through friction on faying surfaces of structural parts, prevents relative motion of parts without the expense of using body-bound bolts. It would seem, therefore, that the production of initial tension is a rather general and important problem in the assembly of bolts and screws.

The most common method of setting tension in bolt or screw shanks is by use of a torque wrench. The problem of determining what tension we have, however, is not too simple. Our benign efforts to produce tension are opposed by some very devilish fellows—friction. We do not know too well, either, where these frictional forces operate or just what their values are, but we do know that the effects of friction are erratic and neutralize a major portion of our effort—as much as 90 per cent or more.

The torque which we apply to a nut or a screw head is absorbed in three general ways. First, there is a frictional resistance on the bearing surface of the fastener which increases with increasing tension and which accounts for some 50 per cent of the applied torque. Second, there is a similar frictional resistance on the bearing faces of the screw threads which accounts for some 40 per cent of the applied effort. Third, there is the useful torque equivalent of the tension produced in the fastener through the screw-jack effect of the thread helix which accounts for a small amount of the applied torque—possibly 10 per cent or less. It is apparent, therefore, that the useful effects of our efforts are relatively minor in character, most of the applied torque being absorbed by indeterminate friction.

It can be demonstrated mathematically that the relation between torque and total induced tension force is expressed as follows:

$$\frac{PL}{W} = u_B R_B + R_T \left[\frac{u_T \sec B + \frac{1}{2\pi NR_p}}{1 - \frac{u_T \sec B}{2\pi NR_p}} \right] \quad \ldots\ldots\ldots (1)$$

Where, P = force applied to wrench (lb)
L = effective length of wrench (in)
W = total induced tension force in bolt or screw shank (lb)
R_p = pitch radius of thread (in.)
R_B = Effective radius of action of frictional forces on bearing face of bolt, nut, or screw (in.)
R_T = effective radius of action of frictional forces on thread contact faces (in.)
u_B = effective coefficient of friction on bearing face
u_T = effective coefficient of friction on contacting surfaces of thread flanks
B = angle between mating faces of threads and a normal to the thread axis
N = number of threads per inch

In equation 1, the values of R_p and N are definite and known for any thread form and can be taken from standard thread tables. An evaluation of the remaining variables should take into account the following considerations: u_B will depend not only upon the surface condition of bearing faces of bolt, nut, or screw, but also upon the surface condition of work parts upon which they bear. These surfaces may be forged, rolled, or cast; may be machined to various degrees of smoothness; may be of various materials; may be coated or plated in various ways; may be dry or lubricated to a more or less extent. As a result, u_B may have any value from 0.05 to 0.40 or more.

R_B also is quite variable. As a standard for comparison it might be assumed that the bearing face is in contact with the faying surface throughout, that bearing pressures are uniform, that the coefficient of friction u_B is uniform over the contacting surfaces, and that the bearing face is circular (washer-faced) having an outer radius R_o, and an inner radius R_i. On these assumptions it can be shown mathematically that

$$R_B = \frac{2}{3}\left[\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right] \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

A close approximation to the above equation for normal proportions of bearing faces of bolts and nuts is the average of R_o and R_i. This equation, however, is based upon certain assumptions that are not always realized in practice. Bearing faces, even when washer-faced, do not always bear evenly on faying surfaces (See Figure 1). If contact tends to be toward the outer radius of the bearing surface because of concavity of mating surfaces, then R_B will approach R_o. If mating surfaces are slightly convex, then R_B will approach R_i. (Note: As loading increases, the bearing face of bolts or nuts may tend to bend slightly and become somewhat convex.) Further, when unfinished nuts (or bolts where torque is applied to bolt) are used, there is no nicely-prepared, circular washer-face and the extreme corner of the hex or square face may drag, thus tending to increase the value of R_B. All of these variations, of course, tend to neutralize themselves (except bending of washer face) as loading increases because of the better seating of bearing surfaces. These variations may be as much as plus or minus 25 per cent of the normal R_B.

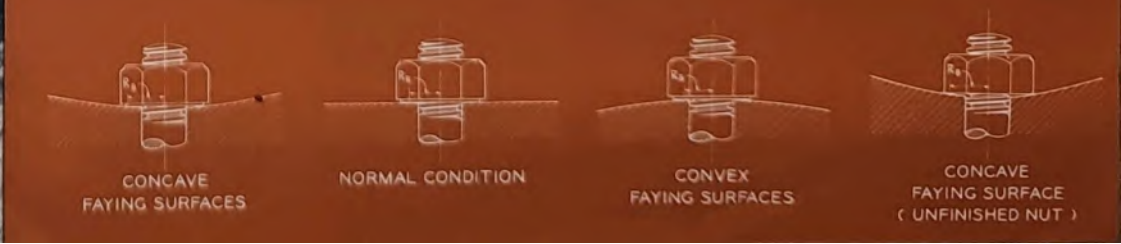
Less variable than u_B is u_T, since threading is a more closely controlled operation than that of preparing bearing surfaces of fasteners or their mating surfaces. Thread surfaces, however, may be produced by cutting, by grinding (which produces a smoother surface), or by thread rolling (which produces a very smooth surface). Further, plating of various kinds may be present, and surfaces may be dry or lubricated. The value of u_T is probably between 0.05 and 0.25.

SecB has a basic value of 1.15 for a thread form angle of 60 degrees. As threads are loaded, however, they tend to bend elastically as cantilevers, thus increasing the value of secB. This effect is slight at low loads, but may be a controlling factor at high stresses. Under very high loads approaching the stripping strength of threads, enlarged photographs show that the angle B may be as much as 45 degrees rather than the normal 30 degrees, in which case secB will have a materiallyincreased value.

R_T is approximately equal to the thread pitch radius R_p but may vary somewhat if thread flanks do not mate uniformly (See Figure 2). With slightly mismating thread form the value of R_T may approach either the crest or the root radius of the bolt threads. This introduces an error of as much as plus or minus 15 per cent for a ¼ inch coarse thread bolt and somewhat less for larger bolts and fine threads. This variation may tend to reduce, and the value of R_T to approach its normal value of R_p, as load increases and threads seat themselves better.

Inspection of equation 1 shows that for a given bolt and nut the relation between torque and tension should be proportional and this is pretty much true at low loads. At high loads, however, the proportionality does not continue because of bending of threads and possibly other effects. As an illustration, Figure 3 shows a typical curve of tensile stress in a bolt vs. applied torque. This is a ⅜ in.-18 plated bolt of about 125,000 psi. tensile strength and about 100,000 psi. yield strength. It will be noted that direct proportionality between stress and torque continues up to a stress of about 73,000 psi.,

FIGURE 1



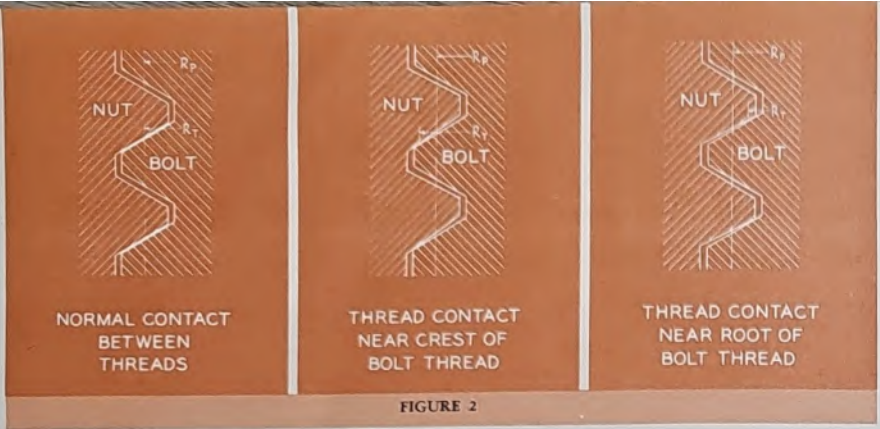
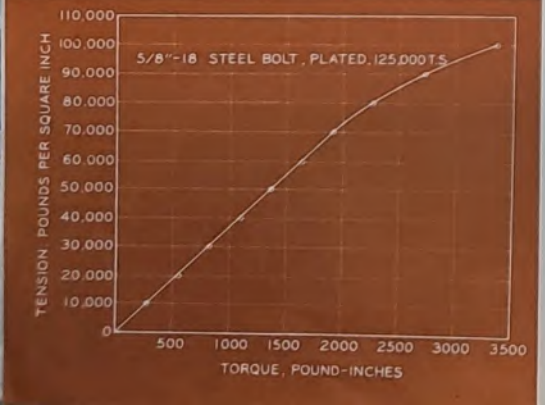


FIGURE 2

beyond which it takes increasingly more torque to produce increasing stress. This limit of proportionality will vary with the size of the bolt and other factors, and does not indicate that the bolt is approaching failure but merely illustrates the changing relations at high stress.

The conclusion from the above discussion is that equation 1 gives a basic value of torque-tension ratio, but the variations from this basic value may be considerable. This should dispel the fairly common assumption that a bolt-nut combination is a simple machine having a fixed relation between applied torque and induced tension. Such a threaded connection is a very inefficient torque-tension machine—in the neighborhood of 10 per cent efficiency. Friction losses are a major factor in the torque-tension ratio and are highly variable. It is too much to expect, therefore, that a precise universal table of torque-tensions can be constructed that will accurately predict the tension in a bolt resulting from the application of torque to the nut. Future research, however, may develop a general table of maximum and minimum limits of torque to produce a desired tension, whose range of variation may be narrow enough for general purposes.

FIGURE 3



In lieu of future research which may develop more definite information, it is proper to ask what is the best procedure to use at the moment. The answer is to eliminate as many known variables as possible by a study of each individual application. The fastenings used should be confined to one type as to surface finish and lubrication, and this should be maintained as near constant as possible. Likewise, the surface conditions of work parts on which the fastenings bear should be similarly controlled. The torque-tension ratio measured for this particular application may then be used for assembly purposes with reasonable certainty of getting the desired bolt tension.

From the foregoing discussion, the fallacy is apparent of one assumption that has occasionally been made. In some instances it has been assumed that the torque-tension ratio is very precise. On this assumption it has been specified that purchased bolts must be capable of withstanding some set torque, and bolts have been rejected as weak which fail under this torque. This is not a correct assumption since the frictional conditions are so variable that it is more likely that the questionable bolts failed because of chance excess in lubrication which would greatly reduce frictional effects and allow a much larger proportion of the applied torque to be converted into an excess tensile force in the bolt shank. Thus, it is apparent that torque at failure cannot reasonably be considered as a measure of the tensile strength of a bolt.

The torque-tension ratio is a very important factor in fastener assembly for many purposes. In view of the lack of definite information at present as to what this ratio is, the Institute is collecting all available data. Suggestions will be welcomed, especially data on experiences of users of fasteners in particular applications.

Heat Treating

by L. S. COOCH
Metallurgist, Buffalo Bolt Company

. . . heat treating procedures have a vital influence upon product quality and performance . . . progress has been made to a point where heat treatment can be controlled so that all product requirements can be met . . .

THE EVER-GROWING use of steels of high physical properties and the increased use of new alloy steels have made closer control of heat treatment, chemistry, and physical characteristics of the metals involved prime requisites in the expected results of the finished products.

The mills of today are constantly trying to produce steel of the best quality, using every possible bit of information derived through investigation and research in order to enable the fabricator to predict with some degree of certainty the results that may be expected and the difficulties that may be encountered. Information so obtained has been of undeniable help and has made possible the utilization of low alloy steels having high physical properties.

While this information serves its purpose, it is of no use with respect to final results if the heat treating furnaces are not of the best and most modern design. While progress in furnace design has not been phenomenal, it has more than adequately fulfilled the requirements of the most stringent specifications. The furnaces of today are controlled by accurate automatic instruments and have adequate equipment to overcome excessive cracking, distortion and lack of uniformity, and still maintain uninterrupted production schedules.

Heat treating procedures and equipment for bolts are typical of those usually applied to small parts. The equipment discussed below is a key factor in the production of numerous sizes and kinds of bolts with uniform physical characteristics from various grades of steel.

The equipment consists of three important parts, which must function properly, both individually and collectively, to produce high quality products. These parts are the furnace, the gas generator, and the quenching equipment.

The points to be considered in furnace construction include:

1. Type.
2. Simplicity of construction and design.
3. Method of heating.
4. Economical and efficient operation.
5. Flexibility.
6. Cost of maintenance.
7. Delays due to possible repairs.
8. Instrumentation or control.
9. Production.

Figure 1 shows the discharge end of two electric furnaces, the half-circular drum in the immediate foreground being a revolving washing drum which removes excess oil from the product after it leaves the quench tank conveyor. These furnaces are continuous, endless conveyor type, atmospherically controlled and hermetically sealed, with each furnace having its own gas generator, one of which is equipped with a refrigeration unit to remove moisture from cracked gas in order to meet the exacting specifications for decarburization. The furnaces, with a capacity of 1000 lb. per hr. each, are equipped with automatic controls for temperature, and speed adjustment for the conveying belt to accommodate different size bolts. These furnaces, being of an older design, will decarburize to a maximum of .005 in. partial, although this is reduced by refrigeration on one of the furnaces so equipped. This, of course, is not due to the furnace but to the type of atmosphere generator, which makes it impossible to meet any specification that requires no decarburization.

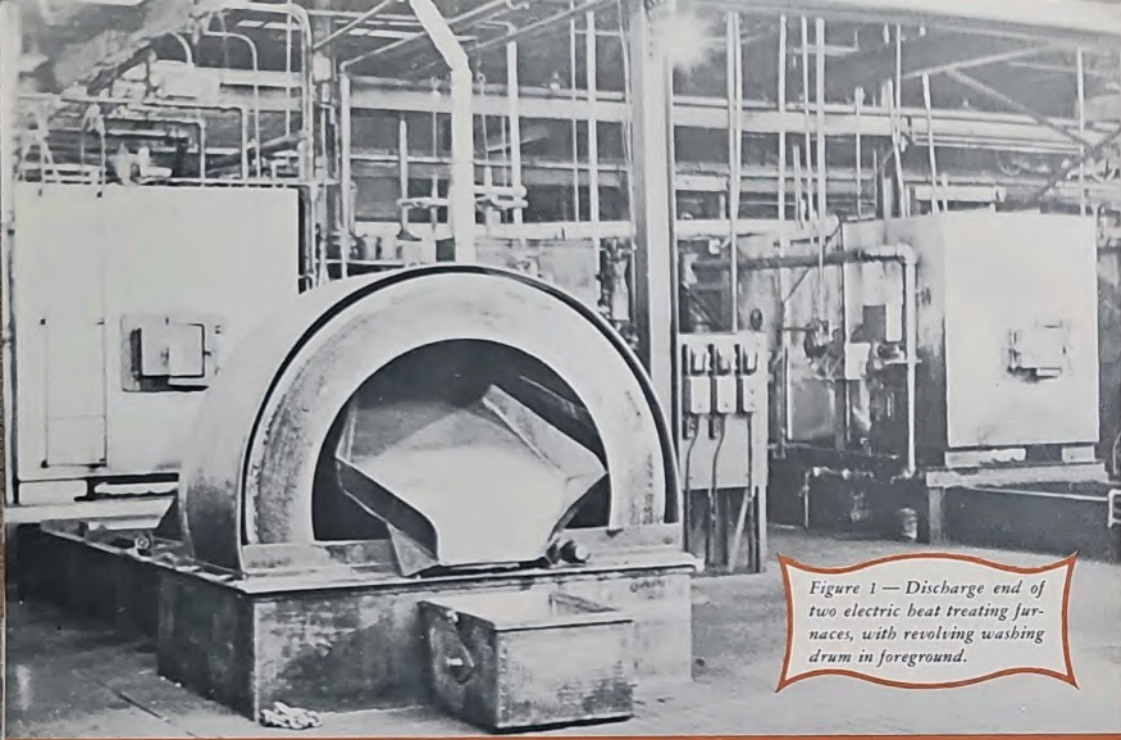


Figure 1 — Discharge end of two electric heat treating furnaces, with revolving washing drum in foreground.

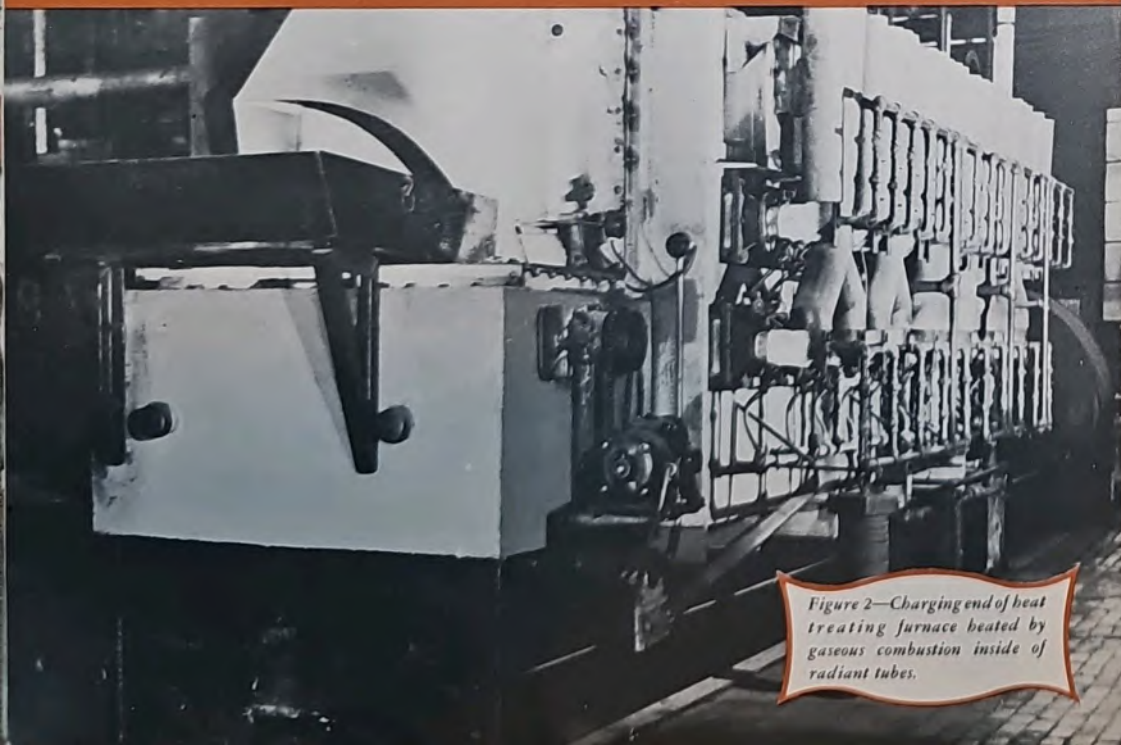


Figure 2—Charging end of heat treating furnace heated by gaseous combustion inside of radiant tubes.

Figure 2 shows a view from the charging end of a more modern type of radiant tube type gas heated furnace. It is hermetically sealed, has a regulated endless conveyor belt and is atmospherically controlled. Hourly capacity is 1500 pounds, compared with 1000 pounds on the electric furnace. This furnace is equipped with 25 heat resisting radiant tubes, the discharge of which is shown on the right side of Figure 2.

The instrumentation of this furnace consists of four pyrometer controllers, two of which are recording and two indicating. One pyrometer controls the temperature of the generator, the other three controlling the temperature of the furnace in the charging zone, the center zone, and the discharging zone.

The major differences in the two types of furnaces are in the rated production and the relative amount of lost furnace time for repairs. The construction of the electric furnace is such that the burning out of an electric element or resistor necessitates the shutting down and cooling of the furnace before the element can be changed, regardless of whether it is below or over the conveyor belt. If the element happens to be under the belt, the entire belt has to be removed in order to make the repairs. This is a time-consuming job, with a great loss of production. On the radiant tube type, in case of a tube failure, the furnace may be kept in operation while the change is made, it only being necessary to slide out the old tube and replace it with a new one. The economical comparison of the cost of heating in the two types of furnaces is, of course, subject to the prevailing rates of fuels in each locality. The production capacity of the gas furnace, being 50 per cent greater than that of the electric furnace, offers a saving in labor, as no additional help is required for the greater output.

While the furnace and its controls are important, equally so is the gas generator which also is a governing factor in the final results. Engineering and construction changes in gas generators have by far exceeded progress in furnace designs. Numerous technical papers have been written on cracked gas compositions and their effect on steels at different temperatures. Generators have been radically changed for the adaption of different gases too numerous to enumerate here. Refrigeration and means of cooling have been perfected to the extent that the smallest amount of moisture in the gas is practically obliterated. Drying towers, where necessary, remove all traces of moisture and CO_2.

The first steps were taken some 15 years ago toward using an atmosphere composed of partly burned gases as a protection for low carbon steels during annealing. Since then, atmospheres and generators have advanced to the point where they may be applied on all steels and at different temperatures, with gratifying results.

The gas generator installed on the radiant tube furnace is separately heated with radiant gas fired tubes using gas of 530 Btu per cu ft and is held at a temperature of 1940 F by automatic temperature control. This differs from the older type generator in that the heating of the generator and the burning of the atmospheric gas is individually controlled. This allows for greater flexibility in temperatures and gas mixtures in the newer type, inasmuch as the older type utilizes the gas mixture for heating the gas chamber and furnishes the protective atmosphere for the furnace from the one mixture.

Table I shows a comparison of the unburned gases from the two types of generators and the raw gas used. The higher air-gas ratio of the old type generator is necessary to maintain combustion in the cracking chamber since this acts as the means of heating the chamber and also the production of the unburned gas for protection. This is readily reflected in the analysis of the gas, which has a relatively high CO_2 content that will decompose at high heats. Hence it is necessary to use a refrigerating unit and a drying tower to remove moisture and CO_2.

The accompanying micro-photographs are representative of production from the different types of furnaces and generators. Figure 3 is taken from the electric furnace with the old type generator and shows partial decarburization, which is the best to be expected, although the pieces are scale-free but dark in surface color. Figure 4 is from the same grade of steel, but from the newer type furnace and generator, and shows a neutral surface condition

TABLE I

	Raw gas	Unburnt gas from radiant tube heated generator	Unburnt gas from old type generator
CO_2	1.1%	.2%	4.3%
O_2	1.0	.0	.0
H_2	55.1	53.5	16.7
CO	5.6	16.2	8.7
CH_4	29.1	2.3	.4
N_2	5.1	27.8	69.9
C_2H_4	.3	....	....
Ill.	2.7	....	....
Air-gas ratio	....	.72 to 1	2.85 to 1

Figure 3—Microphotograph showing partial decarburization of product, which had a dark, scale-free surface.

Figure 4—Microphotograph showing no decarburization of product, which had a bright, scale-free surface.

Figure 5—Microphotograph showing heavy decarburization caused by change in gaseous atmosphere. Product nevertheless had a bright, scale-free surface.

which meets the more stringent specifications on decarburization. These pieces come from the furnace bright and scale-free.

Since the clean or bright surface of a bolt is no guide as far as decarburization is concerned, it is possible, through readjustment of the burnt gas mixture, to have an entirely bright surface, free from scale, and still show excessive decarburization as indicated in Figure 5 which was run identical to the two examples shown in Figures 3 and 4 with the exception of a change in the gas mixture. This piece shows a carbon-free area of .005 in. from the crest of the thread.

When purchasing raw rods or wire for bolts today, it is impossible to expect any supplier to accept a specification limiting decarburization. In cases where a "nil decarb" is specified on the finished product, the fabricator is handicapped before he starts, unless he has some corrective measures available to overcome decarburization. Corrective measures are available in the new type radiant tube furnace because it is possible to carburize sufficiently to cover any decarburization that may be on the rods or wire. This is accomplished by the introduction of a 10 to 15 per cent mixture of propane with the atmosphere gas. The advantage of this type of furnace is here again proven over the electric type, wherein it is impractical to introduce propane because of carbon deposits shorting the resistors and causing costly delays. The alloys used in the radiant tubes and the conveyor belt are resistant to the propane atmosphere and offer no problem even during long periods of operation with the gas.

While the furnace and protective atmosphere are of major importance, equally so is the quenching oil and equipment, because they directly reflect in the quality of the finished product. The tank should be of sufficient depth to allow for the proper quenching of the particular product. The equipment should include cooling coils, heat exchangers, or cooling towers, to maintain the oil within specified temperatures under 24 hour operation. Some means of creating a violent agitation of the oil should be an integral part of the tank to obliterate as much as possible localized heating of the oil.

Motor driven propellers are the most widely used method for agitating the oil, and they should be installed far enough below the level of the oil to cause a rolling effect in the oil, rather than a splash in the immediate vicinity of the propeller. Where a splash is encountered, the oil has a tendency to absorb air and foam appears on the surface, tending to retard the quenching rate and to promote oxidation and decomposition of the oil. An extremely high rate of circulation is obtained through proper installation of the agitator and uniform temperature of the oil is effected and maintained.

The oil is circulated to the front of the tank and back along the sides of the tank, passing over cooling coils, then through heat exchangers into a main storage tank. From the storage tank it passes through a continuous filter to remove foreign matter, then back into the quenching tanks which are maintained at proper level and held within specified temperatures. There has been some dispute concerning the advisability of using a diatomaceous earth type filter for quenching oil, the contention

being that it removes the fatty content of the oil, thus lowering the quenching rate. While this contention may be correct for some oils, we have never experienced any difficulty in this respect, but have found that continuous filtering provides a cleaner oil at all times and results in a uniform quenching rate from week to week.

More attention has been given quenching oils in the last few years than ever before and the producers today are furnishing oils to meet the ever-growing demands for oils having a fast quenching speed. There are three qualifications of a quenching oil which are of major importance:

1. The ability to "wet" a hot surface.
2. Uniformity of quenching speed over a wide range of operating temperatures.
3. Maintenance of all these necessary properties throughout the long life of the oil.

Provided the proper oil is purchased, the next step is to use it correctly with respect to circulation, maintenance of certain temperature limits, cleaning or filtering, and quantity required for the system. The ideal quenching oil should be non-oxidizing. It should be an oil that will absorb the heat from metal in the shortest period of time and one that will not break down or form a sludge. It should have the required properties to cool fast above the transformation range and, in order to produce the minimum amount of warping or cracking, it should cool more slowly in the lower ranges where stresses are set up. Although oils may have identical analysis, their performance over a period of time may be vastly different. Analysis is not a decisive item, nor are laboratory tests, which should be taken as only a guide. Inferior oils in use over a period of time will become thick with sludge and will retard the cooling rate, necessitating the discarding of the oil and replacing with new, a costly operation when a large volume of oil is used, to say nothing of the loss of production because of shutdowns required to clean out the system. While it is true that some oils are lower in price, they have not always proved the most economical. Primary consideration should be given as to whether it is a special purpose oil with stability and not just an oil in which to stick a hot piece of steel.

Figure 6 and 7 show a comparison of identical samples quenched under identical conditions but in different quenching oil in use over a period of time. In Figure 6 the piece was quenched in an oil which had been used consistently for seven years without signs of breakdown. The sample shown in Figure 7 was quenched in an oil used for only six months, after which it was necessary to discard the oil and clean out the sludge which resulted throughout the entire system from the breaking down of the oil. The sample quenched in good oil showed a Brinell hardness of 477 while the sample quenched in the inferior oil tested only 269.

Figure 6—Microphotograph showing satisfactory structure in piece quenched in oil which had been in use for 7 years.

Figure 7—Unsatisfactory structure in sample identical to that of Figure 6, but quenched in an oil which had been in use only 6 months.

DISCUSSION ON

How Tight Should a Bolt be?

. . . the paper of this same title, written by J. O. ALMEN and published in "Fasteners" No. 1 and 2 has provoked much discussion, some of which is presented here

by

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IN THE DISCUSSION of simple machine elements, such as threaded bolts, studs or nuts, there is often much confusion. Mr. Almen's paper in question should clarify this.

Highly stressed bolts and nuts on critical applications, and particularly on those of all types of vehicles which must stand up under the severe punishment in the field, are always a matter of serious consideration to those responsible for safe assemblies.

The basic requirements for correct tightening of bolted joints have made the industry conscious of the tightening torque. Although the torque-tension ratio varies and therefore cannot be considered an accurate basis for producing uniform tension in every bolted assembly of the same thread size, the use of torque wrenches does not give as much trouble as might be expected from a first glance at the result of laboratory tests.

The chief offenders are evidently those variables or combinations which affect the friction and over which we have relatively the best control, namely, lubrication and plating. Lubricants with an abnormally high anti-friction value can be avoided at the place of original assembly as well as in service. The effect upon friction of certain plating combinations can be approximately established and a torque allowance made. Since the majority of fastener applications do not require stressing too close to the elastic limit, a sufficiently safe wrench torque can be determined in most cases after lubricating and plating conditions are known. The wide use of torque wrenches on assembly lines as well as in the field, at a time when experienced mechanics with "wrench feel" are scarce, would bear out such a statement.

It may be mentioned in this connection that practical tests with many bolts and nuts have proven that by deliberate use of a special lubricant with a high anti-friction value, the range of irregularity of torque-stress ratio is much reduced. This would represent a practical way to make the use of a torque wrench more reliable.

The value of locking devices in connection with highly stressed and dynamically loaded bolts and studs is questioned by the author in his paper. We, of course, fully agree with this theoretical reasoning as long as the loss of original bolt tension is due to no change in original position of the nut relative to the bolt. A commercial locknut of the best friction type will maintain its position when properly tightened, despite severe vibration and shocks. If these forces tend to embed the bearing face of either bolt head or nut, some of the original bolt tension is lost and the fastener becomes subjected to stress changes, causing fatigue. Locknuts of known types certainly cannot prevent this.

However, the safety aspect of the function of a locknut from then on becomes much more important than the author was willing to admit. After

the bolt is thus deprived of its correct tension, the developed stress range of the operational load certainly will not cause instant failure. A common nut may gradually unscrew completely, which, in certain cases, may cause the assembly to fall apart, whereas the locknut will maintain its position on the bolt. The life of the fastener then, even when under severe fatigue stresses, may still be of considerable duration.

The tremendous demand for locknuts for ordnance, as well as for aircraft, substantiates the safety value of the friction type locknut. The amount of work and consideration which has been devoted to the specifications for this type of nut indicates that a locking feature which permits the repeated use of such a locknut is not only helpful, but quite evidently of paramount importance for the safety of an assembly.

by
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MR. ALMEN is to be complimented on a lucid and valuable contribution to one aspect of the tremendously complicated subject of bolted joints. The writer is completely in agreement with the author's emphasis on the importance of proper tightening of bolted connections.

In Figure 12 the author shows a wide variation in torque-tension ratio with different types of lubrication. The writer has made a number of tests of $\frac{3}{8}$–24 N F 3 assemblies using several types of lubricant, differing from the author's, and found much less variation in friction. The results indicated coefficients of friction of .21 to .23 for clean surfaces and .15 to .17 for various types of lubrication including cadmium plate, soluble cutting oil, chlorinated fatty oil and oxide black with mineral oil. The author's Figure 12 indicates a maximum coefficient of friction of about 1.3 and a minimum of about 0.12. It is considered that, disregarding chemically cleaned surfaces, that the coefficient of friction is substantially constant for all "greasy" surfaces.

This leads to the conclusion that a torque wrench is a much more accurate method of nut tightening than is indicated by the author's treatment.

With regard to the effectiveness of locking devices, our experience indicates that increased bolt loading not only improves the fatigue strength of the bolt, but also increases the resistance to loosening of the nut under severe vibration. We feel that the principal cause of nuts coming loose is usually a cyclic variation of the bolt load. Under these conditions, since the base of the nut carries most of the load, the load variations cause an alternate expansion and contraction or "panting" of the nut at the base. Since there is a relative radial movement of the bolt and nut threads, the resistance to the tangential movement is greatly reduced, and consequently the nut works loose, under the influence of the wedge action of the thread.

All of the effective lock nuts of the friction type provide an elastic clamping pressure on the bolt threads at the top of the nut. The radial movement of the top of the nut under cyclic loading is much less than at the base, so that the friction lock at this point is undisturbed.

The most valuable uses of lock nuts are in those applications where the variation of bolt load is necessarily large, or where the bolt load is small and vibration is severe.

In the assembly of dynamically loaded assemblies under laboratory or manufacturing conditions, it may be possible to adequately control the tension. However, under field conditions, the added safety of an assembly made with a lock nut is highly desirable.

by
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Assistant to the President
Elastic Stop Nut Corporation of America

BY POINTING out the importance of the elasticity of the bolt and the rigidity of the bolted members in the functioning of threaded connections, Mr. Almen has rendered a service to all engineers and designers.

Some misconceptions, however, have arisen regarding Mr. Almen's paper because due cognizance has not been taken of the fact that it deals only

with the problem of bolt failure due to fatigue and not with the general nature and types of failure that may take place in bolted assemblies.

Failure of the bolt in fatigue is obviously only one form of possible failure of a bolted connection and in many practical instances, may be the least important. Frequently, the problem of far greater seriousness is that of insuring that the assembled parts will remain secure and that the nut will not turn in relation to the bolt under any conditions encountered in actual operation.

In this article, Mr. Almen disregards the possibility of nut loosening. As a result, he fails to appreciate the vital importance of self-locking devices. His statement that "a nut that has loosened in operation does not need a lock because it is no longer performing its function" fails to reckon with the fact that the attachment of a self-locking device to the nut might have prevented the very condition that he describes from taking place.

It is precisely because ordinary nuts do loosen in actual practice, that self-locking nuts are today in such wide demand. Generally speaking, their use offers the engineer or designer the following advantages:

(a) They provide an inherent "factor of safety" in the form of positive resistance against nut movement even when the structure of which the bolted assembly is a part is subject to dynamic or shock loading.

(b) They permit of ready adjustment to any desired degree of initial tightness (Self-locking nuts of the friction type generally offer a resistance to bolt rotation of only about one-tenth the generally recommended values of tightening torque. It is therefore, difficult to correlate the author's statement that self-locking nuts of the friction type "add to the difficulty of attaining proper nut tightness" with his own contention that torque wrenches are of little use because "the friction between the threads of the bolt and the nut is so variable that the bolt tension will vary over wide limits.")

(c) They give positive locking in any position of the nut on the bolt.

(d) They compensate partly for the imperfections and deficiencies in the component parts of the assembly by supplying a source of holding power that is independent of axial seat pressure and of the nature of the contact surfaces.

The great utility of self-locking nuts becomes even more evident when it is recognized that—even where great attention is paid to application conditions and the bolts and nuts used are of the very best quality—it is impossible to guarantee that the ordinary threaded connection will remain secure. This lack of certainty is due in the first place, as Mr. Almen himself points out, to the fact that "no adequate means are available for determining nut tightness except when the elongation of the bolt can be measured." Since this latter is seldom a practical possibility, it is clear that there will be considerable latitude in the initial tightness of bolted connections even where torque wrenches are used. Secondly, operating conditions are frequently so unpredictable that even a properly tightened ordinary nut-bolt combination may be caused to loosen by reason of the forces arising from relative vibration of the parts.

For example, it is well known that a heavy weight, statically resting on an inclined plane and prevented from sliding down by the more than ample frictional forces resisting motion, will easily move when the plane is caused to vibrate by a slight tapping force. In like manner, dynamic operating conditions may materially reduce the frictional forces that exist between the nut and bolt threads and between the nut and assembly seating surfaces. When this happens, the pressure created by the wedging action of the threads tends to cause relative motion of the nut on the bolt.

That this condition actually takes place can hardly be debated as it is common knowledge based on many years of experience that an ordinary nut will frequently loosen and turn even when pressed tight against an assembly. The actual use in practice of the cotter pin and the very existence of thousands of self-locking nuts of one type or another is adequate evidence of this fact.

Some of the past failures that led engineers and scientists to develop a new type of fastening with a self-contained locking element may have been due to improper pre-stressing but, since proper pre-stressing is, as the author points out, usually unattainable, this fact is an argument for rather than against the use of self-locking nuts.

The loosening of a threaded connection—whether due to improper pre-stressing or to conditions of vibration—may not be a very frequent occurrence and in a given instance, may happen to only one nut out of a large group. Yet the results of such a happening might be quite catastrophic. The loosening of one nut means that any load which this nut-bolt combination was designed to carry must now be borne by other nuts already bearing their full design load. Such a condition could easily result in a structural failure of the entire assembly and in possible severe damage as well as loss of life.

It is precisely because of these possibilities that good designers insist upon extra insurance against nut loosening by demanding that self-locking nuts, in which a locking force independent of seat pressure is always present, be used in place of ordinary nuts.

Fasteners Digest

Structural joints fabricated by riveting have proved highly satisfactory in service. Although one of the factors that determines the strength of a riveted joint is the strength of rivets in shear, it is highly desirable to have enough tension in the rivets to enable them to carry the load by virtue of the friction between the plates, thus relieving the rivets of shear and the accompanying flexure.

Professor Wilson describes tests made upon short deck plate girders from bridges built by the Nickel Plate Railroad about 1900 to determine what tension the rivets retain over this period of forty years service. The tests indicated that many of the rivets retained tension in excess of modern maximum working stresses.

"Do Rivets Retain their Initial Tension?", by W. M. Wilson. Civil Engineering, October, 1944.

Anglo-American-Canadian Screw Thread Conferences — The Combined Production and Resources Board has just released a Report of Conferences on Standardization of Screw Threads and Cylindrical Fits. This is a summary report of conferences in August, 1944, between a Joint U. S.-Canada Screw Thread Mission to the United Kingdom to confer with British representatives on the subject of uniform standards for screw thread design.

Copies of the report may be obtained from the Superintendent of Documents, U. S. Government Printing Office, Washington, D. C.

A whole new set of fasteners and fastening accessories has been developed by the aircraft industry in the past few years. Made to meet the requirements of resistance to shock and vibration and of high speed assembly, these new fasteners have characteristics that every industry can put to work, and represent real improvements in fastening techniques. Illustrated and discussed are blind rivets, bolts and screws of special design, locknuts, lockwashers, speed nuts, self-locking nuts, cage nuts, wing nuts and inserts, as well as several quick-opening devices.

"All Secure". Modern Industry, September 15, 1944.

Development engineers have much upon which to draw in making postwar products better and different, and assembly parts will play an important part in improving product design, sales appeal and cost factors. This article surveys the various fastening components, listing their characteristics and assigning them to the type of parts for which they are suitable fastening agents. Items discussed include standard screws, studs, set screws, rivets, blind rivets, lock nuts, lock washers, speed nuts, staples, eyelets, drive screws, self-tapping screws, etc.

"Better Assembly Parts Affect Product Design", by W. K. Walthers. Electrical Manufacturing, September, 1944.

There has been a marked swing toward the use of the rolled thread in place of the ground thread for high precision work. To date this trend has been largely in the aircraft industry but indications point to a wider application of the technique in post-war production, in spite of the care required in precision work. Some new types of machines have appeared in past years and others are about to be introduced. In general, the machines are of reciprocating types, using flat dies and driven through geared crank or through hydraulic piston, or of rotating types, using two or three rotating cylindrical dies.

This article also discusses the factors entering into rolling thread in high strength alloy steel with tolerances corresponding to class 3 and 4 fits. Die materials and design factors are considered, as are methods of controlling blank diameters.

"An Appraisal of Precision Thread Rolling", by Frank J. Oliver. Iron Age, November 2, 9 and 16, 1944.

Using multiple hydraulic riveters designed for heavy duty, high speed production, "gang riveting" has saved manhours and held rivet rejections to a minimum, as well as eliminating rivet gun marks on surfaces, skin waviness and swelled rivets between layers of skin, in the fabrication of subassemblies at the Ryan Aeronautical Company. The machine is self-compensating for changing stock thicknesses and the amount the rivet head is flattened is also controlled. The hydraulic system which operates the machine develops force sufficient to head a group of aluminum alloy rivets ranging from twenty-four $\frac{3}{32}$ in. rivets to four $\frac{3}{8}$ in. rivets.

"Multiple Riveting of Aircraft Parts", by J. E. Cooper. Iron Age, August 24, 1944.

USE HEADED AND THREADED
FASTENERS
FOR ECONOMY-RELIABILITY